May 21, 1957 W. S. BRUCKER 2,792,732

ROTARY IMPACTING NUT RUNNING TOOL

Filed Sept. 23, 1955 2 Sheets-Sheet 1

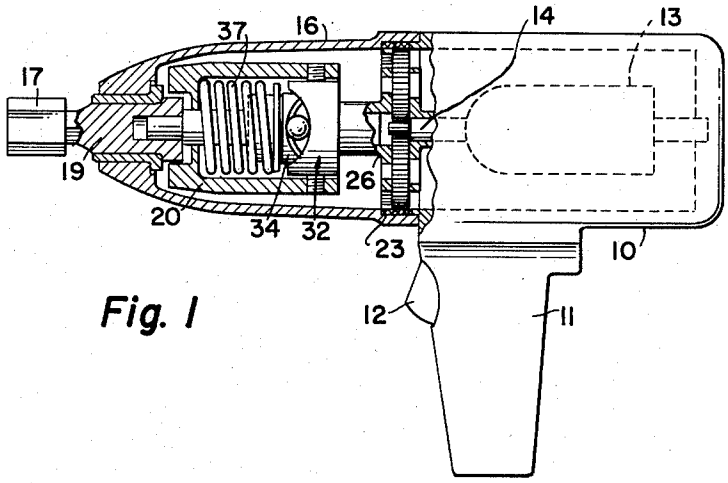
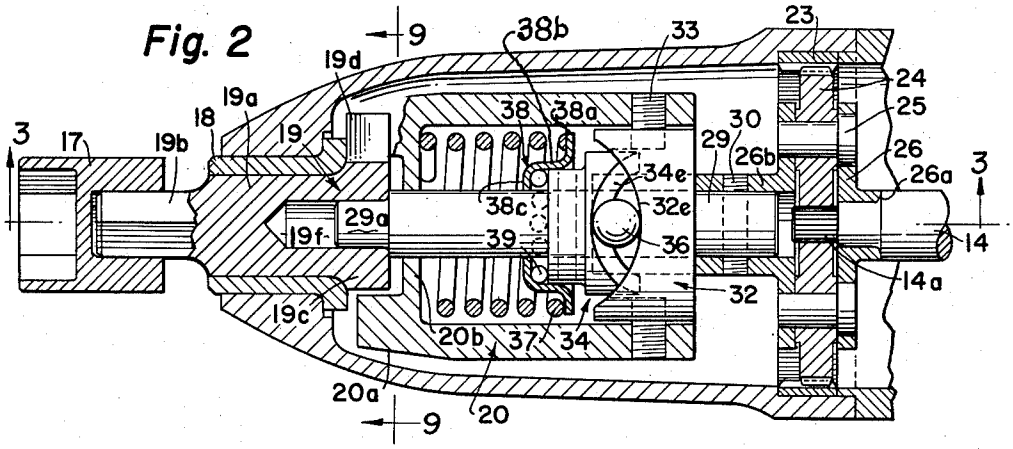


Fig. 1

Fig. 2



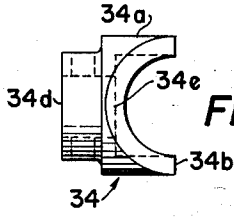


Fig. 4

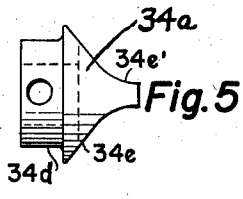


Fig. 5

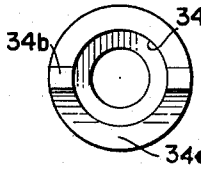


Fig. 6

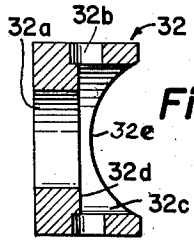


Fig. 7

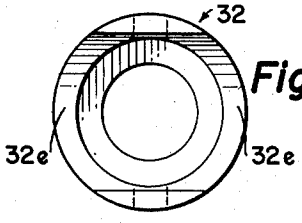


Fig. 8

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ROTARY IMPACTING NUT RUNNING TOOL

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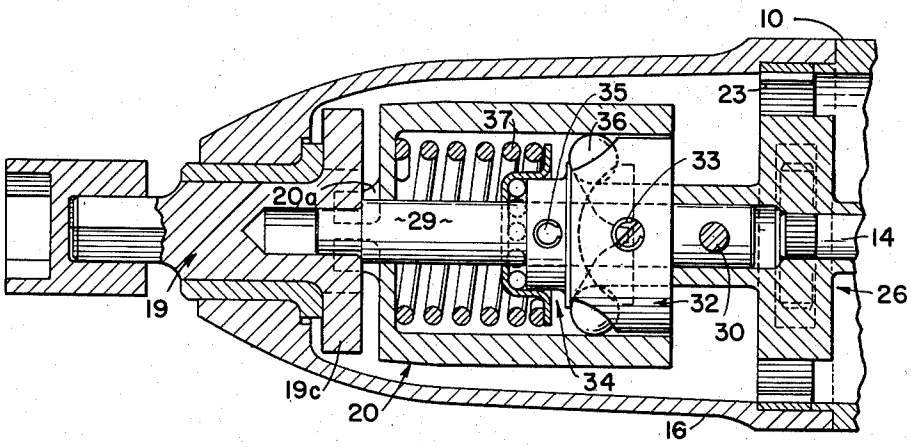


Fig. 3

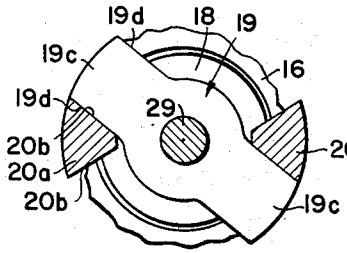


Fig. 9

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2,792,732

ROTARY IMPACTING NUT RUNNING TOOL

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Application September 23, 1955, Serial No. 536,125

6 Claims. (Cl. 81—52.3)

This invention is concerned with impacting nut runners of the compression spring type wherein the connection between an accelerated hammer and the driving means comprises cams and balls.

In tools of this character, the practice has been to connect the hammer to a driving spindle through bearing balls which ride in V-shaped grooves formed on the hammer and driving spindle, whereby the relative motion taking place between the continually driven spindle and the hammer at the instant of nut seating effects a cammed separation of the impact lugs of the anvil and hammer by raising the hamer and its lugs out of clutched relationship with the stalled anvil lugs. Usually such camming moves the hammer axially against a compression spring, and as the hammer lugs rotate past the anvil lugs, the hammer is restored by the spring to a position where its lugs on continued hammer rotation will again engage the anvil lugs in rotational impact, the camming and impact cycle being then repeated. The practice has been to use helically shaped cams, i. e. cam formations with a constant lead, with the result that at the instant of arrested rotary movement of the anvil lugs, therefore of the hammer, axial hammer motion theoretically must start at its full velocity or rate per unit of relative spindle—hammer angular displacement, or time rate velocity with assumed constant spindle speed, and the axial acceleration of the hammer approaches infinity, a parctical impossibility. The actual occurrence entails shock on the driving spindle, cams and balls resulting in friction and heat generation and wearing stresses when the tool is in constant use. Dynamically, when the hammer lugs have impacted on the anvil lugs, some recoil of the hammer relative to the anvil takes place, and an aggravated condition of shock occurs due to the helical shape of the cams.

By the present invention and discovery, the foregoing unfavorable operational action of the cams can be overcome by reducing the initial rate of acceleration of the axial movement of the hammer at the initiation of its separating movement relative to the anvil.

The general purpose of the present invention therefore is the provision of a tool of the type in question which will operate during protracted periods of use without the generation of excessive shock on the hammer driving spindle and cams of the tool.

Another object of this invention is the provision in the impacting mechanism of a pair of co-operating cam formations which will effect acceleration of the axial movement of the hammer in a gradual manner thereby substantially to eliminate shock to the driving elements upon initiation of upward movement of the hammer.

Another object is the provision of a cam mechanism adapted to a more efficient transfer of energy from a driving motor to a spring element intended to accumulate energy to be released in rotational impact blows.

A further object of the present invention is the provision of cam formations and camming mechanisms for the above stated purposes which can be manufactured economically.

A still further object of the present invention is the provision of cam formations having camming surfaces which will be relatively flat with respect to the curvature of the bearing balls therebetween, thereby to avoid the use of ball grooves which per se are friction producing because of extended ball contact.

Other objects and advantages will appear from the following description and drawings, wherein:

Fig. 1 is a side view of a motor driven impact wrench involving the present invention shown in partial axial section;

Fig. 2 is an enlarged detail of the impacting mechanism in the axial section of Fig. 1;

Fig. 3 is an axial section taken substantially along the line 3—3 in Fig. 2;

Fig. 4 is a detail elevation of a cam element carride by the spindle of the impact mechanism;

Fig. 5 is an elevational view taken at right angles to Fig. 4;

Fig. 6 is an end view of the camming surface corresponding to Fig. 4;

Fig. 7 is an axial section of a second cam element carried by the hammer of the impact mechanism;

Fig. 8 is an end view of the element of Fig. 7 showing the camming surface thereon; and

Fig. 9 is a detail view showing the relation of the impact lugs.

In Fig. 1, there is shown, as an electrically driven tool, a portable impacting nut running wrench incorporating the present invention, including a motor casing **10** with attached handle **11**; a trigger type motor switch **12** in the handle; an electric motor within the casing **10**, preferably reversible, with armature or rotor **13** and motor shaft **14**; a deep cup-like impact mechanism housing member **16** with inner end counterbored to receive in close-slip fit a reduced shouldered end of the motor housing, and within housing **16**, a mechanism coupling the motor shaft to tool element **17**, as hereinafter described in detail, adapted to convert energy and continuous rotary motion available at the motor shaft under certain conditions into rotational impact upon a work piece, for example a nut or bolt engaged by a suitable tool element such as the socket wrench element **17**.

Any suitable motor may be used, but as stated preferably a reversible electric motor with reversing switch means so that the tool may be used either for tightening or loosening fasteners such as nuts or bolts. The housing elements **10** and **16** are secured together by such means as screws passed radially through the counterbored end of **16** and threaded in the shoulder portion of member **10**, or screws parallel to the axis securing radially projecting adjacent corresponding integral lugs on those members.

At the convergent left end of the housing member **16**, a coaxial central bore or aperture, aligned with the motor shaft axis, is fitted with a sleeve type bearing **18**, flanged at its inner end for retention in the housing, to journal a shaft portion **19***a* of an anvil stud element **19** projecting through the housing to carry the work engaging tool element **17** on the outer end portion **19***b*, shown square in cross section to fit in a rotational driving relation the complementary recess of the tool element. The inner end of element **19** is enlarged to form integral angularly spaced anvil lugs **19***c*, symmetrical in individual form and relative to the tool axis and in diametric relation to each other, having impact surfaces **19***d* parallel to the axis of the tool. The four impact surfaces **19***d*, roughly approximating but not in precisely radial disposition, for facility of manufacture are parallel to each other and are continued inwardly to merge through a filet into the central or body portion of **19**.

Interposed between the counterbore shoulder at the right end of housing member **16** and the reduced end

of motor housing 10, there is a ring gear element 23 fixed against rotation by press fitting or other means, with which mesh a plurality of planet gears 24 rotatable on stud shafts 25 fixed in a planet carrier or cage 26. The cage is rotatably supported at one side by its central integral tubular extension 26*a* on the relatively rotatable motor shaft 14, the latter having a gear formation 14*a* cut on the end thereof as a sun gear meshing with planets 24; and at the other side by a spindle 29 with one end pinned at 30 in a second tubular extension 26*b* and having the opposite reduced end 29*a* journalled in a coaxial bore 19*f* of anvil 19. The spindle 29 may however be an integral shaft extension of the planetary cage rather than secured thereto in the manner shown.

The shaft 14 has a shoulder outward of the end of cage extension 26*a*. Thus with rotor shaft bearings holding the shaft against any appreciable axial shift, the shaft shoulder bearing on the end of extension 26*a* and the shoulder at reduced spindle end 29*a* bearing on the anvil, the spindle, spindle cam 34 and planetary gearing assembly are held within housing 16 against any appreciable axial movement. The spindle 29 and cage extension 26*a* may be rotatably supported by the anvil and motor shaft and held against axial shift either directly as shown or through known bearing devices of apt form. The planetary system thus affords a reduction gearing between the motor and the driving spindle 29 of the impact mechanism disposed coaxially with and between motor shaft 14 and tool element 17.

A hammer member 20, of generally hollow circular cylindrical shape, is supported for some degree of axial and rotational displacement relative to drive spindle 29 upon operation of the hammer lift camming mechanism 32, 34, 37, 38 during impact action, by its end wall 20*a* coaxially apertured for rotational and sliding bearing on the spindle; and through the cam mechanism hereinafter described, by a symmetrical hollow hammer lift cam member 32 (Figs. 7, 8) of circular cylindrical external form inserted into, and secured against relative axial and rotational displacement in, the open or right end of the hollow hammer and having a bore 32*a* through which with clearance extends the left end of cage tubular extension 26*b*, or the spindle where spindle and cage are integral. Arch-like cam surfaces of member 32 face toward the anvil end of the mechanism. The hammer lift cam member is held with a small circumferential spacing in the hammer by diametrically opposite aligned fasteners 33 serving as trunnions allowing to cam member 32 a limited degree of pivotal movement about the fastener centerline. The fasteners 33 may be dog point set screws threaded through diametrically opposed threaded radial apertures of the cylindrical hammer wall with the dowel-like points of the screws extending as trunnions into and bearing on bottom surfaces of corresponding reamed holes 32*b* in the cam element 32. Snap rings within the threaded apertures may be used to hold the screws 33, and so also to hold pins or screws at 30 and 35. On the closed left end of the hammer element 20, there is a pair of hammer lugs 20*a*, individually symmetrical in form and in diametric relation, extending endwise from the hammer element toward the anvil with impact faces 20*b* parallel to the tool axis disposed complementary to the impact areas of faces 19*d* for engagement and impact with matching impact areas of corresponding faces on the anvil lugs.

A symmetrical spindle lift cam member 34 (Figs. 4–6), having a portion 34*a* with circular cylindric external surface extending into the counterbore 32*c* and end face portions 34*b* having clearance relative to counterbore shoulder 32*d* in the opposed end of cam 32 when the hammer is moved to the extreme left position of Fig. 2, is secured to the drive spindle 29 by a radial pin 35, the cam surfaces thereof facing away from the anvil end in opposed relation to the cam surfaces on 32. A pair of hardened cam balls 36 are interposed between the two

cam elements, each ball lying between corresponding opposed cam surfaces 32*e* and 34*e* of hammer and spindle cam members respectively.

The limited pivotal motion of the hammer cam element 32 about fasteners 33 accommodates larger manufacturing tolerances than would otherwise be possible, particularly in the cams, without adversely affecting the functioning of the tool.

The central aperture of spindle cam 34, through which the spindle 29 extends, is counterbored at 34*c* to a depth beyond surface 34*e* to accommodate the left end of the cage tubular extension 26*b*. A heavy prestressed helical compression spring 37, biasing the hammer toward the anvil, is located within the hollow hammer to bear endwise directly on the hammer end wall 20*b*, and mediately on the spindle cam through a bearing cup member 38 having a radial rim flange 38*a* to engage the corresponding end of the spring, a cylindrical wall portion 38*b* surrounding with clearance a portion of the length of a reduced end 34*d* on the spindle cam as a bearing centering means and a plurality of bearing balls 39 located in the annular space between spindle 29, the left end of cam 34, and the centrally apertured end wall 38*c* and cylindrical wall 38*b* of cup 38 to form a thrust bearing between the spindle and spring in its ultimate function.

In the hammer lift cam 32 of the drawing, the arch shaped camming surfaces 32*e* are symmetrically diametrically formed relative to the cam element axis, therefore to the tool axis—the common axis of the coaxially aligned shaft 14, spindle 29, anvil 19, hammer 20 and spindle cam 34—as portions of a single theoretical circular cylindric surface intersecting the hollow cylinder of the hammer cam with axis spaced beyond the working end of the cam and also perpendicular to the cam axis. Such cam surfaces may be formed on a hollow cylindrical hammer cam blank end at one pass by milling diametrically thereacross with a rotary cutter having suitably shaped cutting edges.

In the spindle lift cam 34 also, the camming surfaces 34*e* are symmetrically formed and diametrically disposed relative to the cam element axis, and hence so located relative to the tool axis. The major portions of these cam surfaces are portions of corresponding theoretical planes equally oblique to the axis and angularly disposed as faces of a dihedral angle, the dihedral edge of which intersects at right angles the axis of this cam element. The axis lies in the bisecting plane of the dihedral angle. However, in approaching the dihedral edge the cam surfaces become curved surfaces 34e^1 diverging from the dihedral faces as appears in Fig. 5, with the terminal portions of the actual cam surfaces approaching parallelism with the cam element axis and forming definite partitions between the two camming surfaces of opposed slope. The working end of the spindle cam is therefore roughly wedge-shaped. The particular constants of the cam surface shapes are chosen to attain curved paths of ball contact therewith such that rolling contact of the balls is ensured.

Thus the spring 37 biases the hammer and therefore the hammer lift cam 32 to the left, toward the anvil, so that the anvil lugs 19*e* would lie in the path of rotation of the hammer lugs 20*a*. The ball elements in the camming mechanism are captive between the inner round cylindrical surface of the hammer, the cam surfaces 32*e*, the cam surfaces 34*e* sloping in a direction in major part away from surfaces 32*e* in a radially outward direction with the relative positions of Figs. 2 and 3, and tubular extension 26*b*, though the wedging action of surfaces 32*e* in spreading the balls apart into contact with the inner surface of the hammer keeps each ball out of contact with the extension for only three point contact with the other noted surfaces under all normal operating conditions. Also with the balls so spread into contact with the hammer, the upper end of the hammer is piloted

or supported relative to the spindle through the balls and rest of the camming mechanism.

With the hammer free for rotation relative to the spindle, the cam elements assume the position of Figs. 2 and 3 under the force of spring 37. Upon rotation of the spindle in either direction by the motor, the hammer lugs contact the anvil lugs as appears for clockwise rotation in Fig. 2 or Fig. 9 to rotate the anvil and tool element 17, the force of the spring being sufficient to maintain the relations of the engaged cam elements as in Figs. 2 and 3 as long as the tool element is not rotationally impeded, for example during the free running of a nut on or off a bolt or stud.

When element 17 is impeded, say upon seating of a nut after free running, or when the tool is applied to remove a seated nut, the hammer is then held against rotation. As the spindle cam is driven further by the motor, effecting a relative angular displacement or advance of cam element 34 relative to element 32 from the "neutral" position of Figs. 2 and 3, the balls are displaced from their positions lowest or innermost relative to the ends of the respective cam surfaces up the cam slopes with ball angular displacement relative to each cam corresponding to substantially equal paths of rolling contact (or to half the relative angular cam displacement for simplified analysis in a non-quantitative sense), displacing the hammer axially away from the anvil thereby compressing spring 37, until the hammer lugs escape the anvil lug impact faces. As such compression is effected, energy is stored in the compressed spring. Thereupon the hammer is free to rotate in the direction of spindle rotation; and, when the end faces of the lugs have cleared each other, for the compressed spring to move the hammer axially to bring the hammer lugs into the space between the anvil lugs for again achieving impact engagement of the lugs at the impact faces. However, as the hammer moves axially forward toward the anvil under spring force, thereby returning the elements of the cam mechanism toward "neutral" position, a relative motion of the hammer to the spindle in the direction of spindle rotation is attained, converting the potential energy stored by the spring compression during the preceding lagging hammer displacement into kinetic energy of the hammer accelerated thereby.

This kinetic energy, additive to the energy supplied by the motor through the rotating spindle upon hammer escape, results in a strong rotational impact force of hammer upon anvil to tighten or loosen a seated nut. The free angular space, represented by the difference between 180° and the sum of the angular extent of a hammer and anvil lug provides time for the acceleration by the motor of the gearing, spindle cam and hammer total mass as well as acceleration of the hammer with respect to the spindle before a consequent impact. Where recoil of the hammer takes place after an impact, the recoil effects a relative motion between spindle and hammer in the same sense as a corresponding initial static engagement of hammer and impeded anvil with attendant lift of the hammer by the cam system, spring compression and accumulation of potential energy in the spring. The potential energy of the spring is of course available for conversion into kinetic energy as previously described, additive to the kinetic energy acquired by the rotating impactor system acceleration before another impact.

The continued rotation of the spindle by the motor causes a repetition of such impact cycle and a rapid succession of rotational blows as usual in tools of this character to effect a desired degree of tightening within the capabilities of the tool. The strength of the spring, its degree of precompression and the power and operating characteristics of the motor are chosen by the ordinary design considerations for the tool capacity required.

Considering the geometrical relations of the camming mechanism in the operation just described, it will first

be noted that the contact path of a ball on a surface 34*e*, i. e. the plane cam surface portion, is a portion of a plane curve, approximately an ellipse oblique to the tool axis and having a major axis passing through the tool axis. Thus at the "neutral" position of Figs. 2 and 3, such ball has a point of contact with surface 34*e* at the end of the major axis, the lowermost point of the path, and lying on a line there tangent to the ellipse and accordingly to the circle described about the tool axis during rotation by such lowermost point, i. e., the end point of the major axis. Therefore at the beginning of relative angular displacement in either direction between the cams, the pressure angle of cam 34 on the ball is zero, in contrast with the considerable angle of a helical cam for such service. Of course, in the case of cam 32, the contact path of the ball on surface 32*e* is not a plane curve, but rather a skew curve, approximately lying in a circular cylindric surface coaxial to the tool and broadly being a portion of the intersection, with the latter surface, of a theoretical second cylindric surface (of which 32*e* is a portion) generated by a generatrix line (defining, at its continuously successive positions, the cylindric elements of the surface) moving perpendicular to a plane including both the corresponding directrix curve and the axis of the tool rotation, the directrix being symmetrical to the axis of rotation. However, the ball has at neutral position a point of contact with 32 lying on a tangent to the low point of the path and accordingly to the circle described by that low point about the tool axis during rotation, so that here too the pressure angle of ball and cam 32 is zero as just noted for cam 34. In the specific form of the drawing, the theoretical cylindric surface corresponding to 32*e* is circularly cylindric. The net effect then at the instant of beginning relative angular displacement in either direction between the cams, since the pressure angles are zero, is that there is not a theoretically necessary axial motion of some finite value instantaneously imparted to the hammer with infinite acceleration; and in a practical sense shock due to such cause is in fact substantially obviated, a result of particular value during the series of repeated impacts involved in a nut tightening or loosening operation. In a practical sense the above analysis and operative results obtain, even though some slight tilt of the hammer lift cam out of coaxial relation with the tool axis occurs for accommodation of manufacturing tolerances.

By inspection it may be seen that from the initial neutral position above analyzed, as relative angular displacement occurs between spindle cam and hammer cam to lift the hammer, the slopes of the ball contact paths thereon increase, so the axial hammer displacement occurs at an increasing rate in terms of relative angular displacement; though as to time rate, as the motor speed drops off with increasing spring reaction, in the latter part of angular displacement, the time rate actually drops off. Hammer axial acceleration in terms of time increases from a relatively low initial value to a maximum, drops to zero and becomes negative as the axial hammer velocity decreases to zero under spring resistance. In terms of the relative angular displacement, the axial motion component imparted by cam 34 is harmonic in quality; and by hammer cam 32, a modification of a cyclic motion.

I claim:

1. In a tool of the character described, a hammer driving spindle, a rotary anvil and a hollow hammer, cooperating impacting lugs on the anvil and on the hammer, said hammer being displaceable in a rotational and axial sense relative to said driving spindle, energy accumulating means comprising a compression spring reacting on the spindle and hammer normally to maintain the hammer and anvil lugs in engagement during nut running operation of the tool, and means for axially shifting the hammer away from the anvil upon arrested movement of the anvil in opposition to the influence of the spring;

the second said means comprising a hammer cam member fixed on the hammer having a pair of arch-shaped cams facing toward the direction of the anvil and disposed 180° apart, a wedge-shaped cam member having a pair of plane cams formed at an obtuse dihedral angle to each other and symmetrically converging toward the axis of the spindle and disposed on the spindle 180° apart, said plane cams facing in a direction away from the anvil and normally in opposed relation to the arch-shaped cams on the hammer, and a bearing ball interposed between each set of opposed cams; said hollow hammer having an internal circular cylindrical surface serving to retain the balls in rolling relation to the said cams.

2. In a tool of the character described, a hammer driving spindle, a rotary anvil, a hollow hammer having one end open and an opposite end wall, cooperating impacting lugs on the anvil and on the hammer end wall, said hammer being mounted coaxially about the driving spindle and being displaceable in a rotational and axial sense relative thereto, energy accumulating means comprising a compression spring within the hollow hammer reacting on the spindle and hammer end wall normally to maintain the hammer and anvil lugs in engagement during nut running operation of the tool, and means located within the hollow hammer for axially shifting the hammer away from the anvil upon arrested movement of the anvil in opposition to the influence of the spring; the second said means comprising a hammer cam member fixed in the open end of the hammer and having a pair of arch-shaped cams facing inward of the hammer and disposed 180° apart, the spindle extending through a bore in said cam member and the cam member being counterbored between the cams, a pair of flat spindle cams shaped to be obliquely symmetrically converging toward the axis of the spindle and disposed on the spindle 180° apart, said flat spindle cams being extended in the counterbore of the hammer cam member normally in opposed relation to the arch-shaped cams on the hammer, and bearing ball means interposed between each set of opposed cams, said hollow hammer having an internal circular cylindrical surface portion serving to retain said ball means in rolling relation between the opposed cams.

3. In a tool of the character described, a hammer driving spindle, a rotary anvil and a hollow hammer, cooperating impacting lugs on the anvil and on the hammer, said hammer being displaceable in a rotational and axial sense relative to said driving spindle, energy accumulating means comprising a compression spring reacting on the spindle and hammer normally to maintain the hammer and anvil lugs in engagement during nut running operation of the tool, and means for axially shifting the hammer away from the anvil upon arrested movement of the anvil in opposition to the influence of the spring; the last said means comprising a hollow cylindric hammer cam member fixed on the hammer with the spindle extending coaxially therethrough, said cam member having a cylindric trough-like formation across an end thereof with the cylindric elements perpendicular to a plane including the spindle axis whereby are formed a pair of arch-shaped cams facing toward the direction of the anvil and disposed 180° apart, a spindle cam member of circular cylindric external shape with one end projecting into the end of the trough formed end of the hollow hammer cam member, said one end of the spindle cam member being substantially cuneate to form a pair of plane cams at an obtuse dihedral angle to each other disposed on the spindle 180° apart and symmetrically converging toward the axis of the spindle, said plane cams and arch-shaped cams being normally in opposed relation, and a bearing ball interposed between each set of opposed cams and having a single point of contact with each cam, said hollow hammer having an internal circular cylindrical surface serving to retain the balls in rolling relation to the said cams.

4. In a tool of the character described, a hammer driving spindle, a rotary anvil and a hollow hammer, cooperating impacting lugs on the anvil and on the hammer, said hammer being displaceable in a rotational and axial sense relative to said driving spindle, energy accumulating means comprising a compression spring reacting on the spindle and hammer normally to maintain the hammer and anvil lugs in engagement during nut running operation of the tool, and means for axially shifting the hammer away from the anvil upon arrested movement of the anvil in opposition to the influence of the spring; the second said means comprising a hollow cylindric hammer cam member fixed on the hammer with the spindle extending coaxially therethrough, said cam member having across an end thereof a symmetrical circularly cylindric trough-like formation with the cylindric elements perpendicular to a plane including the spindle axis whereby are formed a pair of arch-shaped cams facing toward the direction of the anvil and disposed 180° apart, a spindle cam member of circular cylindric external shape with one end projecting into the end of the trough formed end of the hollow hammer cam member, said one end of the spindle cam member being substantially cuneate to form a pair of plane cams at an obtuse dihedral angle to each other disposed on the spindle 180° apart and symmetrically converging toward the axis of the spindle, said plane cams and arch-shaped cams being normally in opposed relation, and a bearing ball interposed between each set of opposed cams and having a single point of contact with each corresponding cam, said hollow hammer having an internal circular cylindrical surface serving to retain the balls in rolling relation to the said cams.

5. In a tool of the character described, a hammer driving spindle, a rotary anvil, a hollow hammer having one end open and an opposite end wall, cooperating impacting lugs on the anvil and on the hammer end wall, said hammer being mounted coaxially about the driving spindle and being displaceable in a rotational and axial sense relative thereto, energy accumulating means comprising a compression spring within the hollow hammer reacting on the spindle and hammer end wall normally to maintain the hammer and anvil lugs in engagement during nut running operation of the tool, and means located within the hollow hammer for axially shifting the hammer away from the anvil upon arrested movement of the anvil in opposition to the influence of the spring; the second said means comprising a hammer cam member fixed in the open end of the hammer and having a pair of concave end cams facing inward of the hammer and disposed 180° apart in rotational symmetry relative to the spindle axis, the spindle extending through a bore in said cam member and the cam member being counterbored between the cams, said end cams being portions of a common cylindric surface having cylindric elements perpendicular to a plane including the spindle axis and each having the axially innermost portion thereof tangent to a plane perpendicular to the spindle axis, a pair of flat spindle cams shaped as portions of planes obliquely symmetrically converging toward the axis of the spindle, the spindle cams being disposed on the spindle 180° apart in rotational symmetry relative to the spindle axis, said flat spindle cams being extended in the counterbore of the hammer cam member normally in opposed relation to the cams on the hammer, and bearing ball means interposed between each set of opposed cams, said hollow hammer having an internal circular cylindrical surface portion serving to retain said ball means in rolling relation between the opposed cams.

6. In a tool of the character described, a hammer driving spindle, a rotary anvil and a hollow hammer, cooperating impacting lugs on the anvil and on the hammer, said hammer being displaceable in a rotational and axial sense relative to said driving spindle, energy accumulating means comprising a compression spring reacting on the spindle and hammer normally to maintain the hammer

and anvil lugs in engagement during nut running operation of the tool, and means for axially shifting the hammer away from the anvil upon arrested movement of the anvil in opposition to the influence of the spring; the second said means comprising a hollow cylindric hammer cam member with the spindle extending coaxially therethrough, aligned pivot means diametrically disposed to the spindle axis securing the hamer cam member to the hammer against relative axial and rotational displacement while permitting a limited self-adjusting pivotal motion therebetween, said cam member having across an end thereof a symmetrical circularly cylindric trough-like formation with the cylindric elements perpendicular to a plane including the spindle axis whereby are formed a pair of arch-shaped cams facing toward the direction of the anvil and disposed 180° apart, a spindle cam member of circular cylindric external shape with one end projecting into the end of the trough formed end of the hollow hammer cam member, said one end of the spindle cam member being substantially cuneate to form a pair of

plane cams at an obtuse diheral angle to each other disposed on the spindle 180° apart and symmetrically converging toward the axis of the spindle, said plane cams and arch-shaped cams being normally in opposed relation, and a bearing ball interposed between each set of opposed cams and having a single point of contact with each corresponding cam, said hollow hammer having an internal circular cylindrical surface serving to retain the balls in rolling relation to the said cams.

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